Consumer and Corporate Affairs Canada
Consommation et Corporations Canada

(11) (A) No. 1 205 654

(45) ISSUED 860610

(52) CLASS 74-191

(51) INT. CL. [4] F16H 3/40

(19) (CA) CANADIAN PATENT (12)

(54) Reversing Rotary Drive Transmission Means

(72) Tschudin, Oswald,
Switzerland

(73) Granted to Institut Straumann AG
Switzerland

(21) APPLICATION No. 436,890

(22) FILED 830916

(30) PRIORITY DATE Switzerland (5589/82) 820922

No. OF CLAIMS 22

Canada

DISTRIBUTED BY THE PATENT OFFICE, OTTAWA.
CCA-274 (11-82)

ABSTRACT

Reversing rotary drive transmission means for converting continuous rotation in one direction into rotation alternately in mutually opposite directions comprises a drive bevel gear (35) rotatable by a drive shaft and two driven bevel gears (51, 53) mounted on a driven shaft (49), the drive gear (35) having teeth on only a sector thereof so as to alternately drivingly engage the two driven gears (51, 53) as the drive shaft rotates. The gear teeth on the driven gears (51, 53) have the same tooth pitch and are offset relative to each other by an arc length measured along the pitch circle which is not greater than one quarter of the tooth pitch.

The present invention relates to reversing rotary drive transmission means and has particular reference to a gear mechanism wherein continuous rotation in one direction can be converted into rotation alternately in mutually opposite directions.

Devices for converting unidirectional rotation into rotation alternately in mutually opposite directions are known in the prior art from, for example, Swiss patent specification No. 610 753 and corresponding U.S. patent specification No. 4 111 208, which disclose different variations of mechanisms for converting the rotational movement of a motor of a drilling machine having a single rotational direction into rotational movement of a drill bit having alternating opposed rotational directions. In a first variant, there is provided a drive shaft with a crank, a slide and a driven shaft which is also provided with a crank. The rotational axes of the two shafts are aligned with one another and the slide is displaceable at right angles relative to these rotational axes. Each of the cranks has a crank pin which is eccentric relative to the shafts and which engages in a bore of a slide part.

In a second variant there is also provided a drive shaft with a crank and a slide. The latter is provided with a toothed rack which meshes with a pinion seated on the driven shaft.

In a third variant, the drive shaft is provided with a head which has an annular groove extending along a plane which is inclined relative to the rotational axis of the drive shaft. A slide, which is guided so as to be displaceable parallel to the rotational axis of the drive shaft, has a pin engaging in an annular groove as well as a toothed rack which

meshes with a pinion attached to the driven shaft.

The conversion of the movement is thus effected in all of these variations by means of a slide which is displaceably guided along a straight path. Since forces which are also directed transversely relative to the displacement direction of the slide act upon the slide, friction losses will increase and the necessity arises for relatively expensive guide means.

Additionally, the slide and the guide means necessary for effecting guiding action result in a relatively large, heavy mechanism. Since such a mechanism is arranged on a hand drilling machine provided for surgical purposes, large dimensions and a heavy weight of such a mechanism make the work of a surgeon utilizing the machine more difficult.

At least in the first of the three variants discussed above, the driven shaft is, in each instance, rotated with bidirectional opposed motion about a rotational angle which, according to structural require-ments, must be smaller than 180°. This produces the disadvantage that special drill bits having three cutting edges must be utilized in place of the usual drill bit having two cutting edges.

If a drilling machine is equipped with a mechanism that is constructed in accordance with one of the three variants discussed above, and if the drive shaft of the mechanism is rotated at a constant speed, the driven shaft will effect opposed bidirectional rotation with the angular velocity increasing in a sinusoidal manner to a maximum value and then immediately gradually decreasing again. As a result of this speed characteristic, a disadvantage arises in that the drill can be operated with a cutting speed which is appropriate for the material to be drilled only during short periods in its alternating rotational

movement.

In German Utility Model No. 1 941 557 there is disclosed a gear device for the conversion of unidirectional rotation of a drive shaft into reciprocating rotation of a driven shaft, wherein the two shafts as well as two intermediate shafts are rotatable about two axes parallel to each other. The drive shaft and the two intermediate shafts are each provided with a wheel having a toothed sector, and the toothed sector of the wheel on the drive shaft comes into engagement alternately with the toothed sectors of the other two wheels. The tooth-free sector of the wheel on each of the intermediate shafts is provided at one end with an edge radially projecting beyond the toothing. The two intermediate shafts also have fully toothed intermeshing gears and one of the intermediate shafts is further provided with an additional gear which engages in a toothed sector of the driven shaft. When the drive shaft is rotated during operation, the toothed sector of its wheel alternately engages at the aforesaid projecting edges of the wheels on the intermediate shafts and rotates these into a setting in which it can engage in the toothed sectors of the intermediate shaft wheels.

It is not indicated in German Utility Model No. 1 941 557 exactly how the toothings of the toothed sectors of the wheels on the intermediate shafts are arranged relative to each other. However, it is reasonable to infer that it would be provided that the teeth of both sectors should have the same positions with respect to phase, i.e. that at least a part of the teeth would exactly overlap if the two toothed sectors were notionally axially projected one on the other. According to the drawing of the Utility Model, on reversal of rotational direction

the toothed sector of the drive shaft wheel has a rotational setting in which it is completely out of engagement, i.e. does not engage in the toothed sectors of the wheels of either of the intermediate shafts. Thus, in this operational phase the intermediate shafts and thereby also the driven shaft can freely rotate at least in that direction in which the afore-mentioned projecting edge of the wheel, which should next come into engagement with the drive shaft wheel, moves on from the start of the toothed sector of the drive shaft wheel. However, it is then not defined how the toothings of the drive shaft wheel and the intermediate shaft wheels are disposed relative to each other, so that there is a high probability that the device is blocked.

In the German Utility Model No. 1 941 557 the use of the gear device is not mentioned in detail. It can be surmised, however, that this device is provided for relatively slow-running control processes. This known device would certainly not be suitable for an operation in which strong counter rotational moments are exerted on the driven shaft. If, for example the device were used for the driving of a drill bit, counter rotational movements could be exerted by the bit on the driven shaft. These would result, on reversal of the rotational direction, in small rotational movements of the intermediate shafts, which for the reasons explained in the preceding would lead to blocking of the gear device.

However, the device described in the Utility Model would also not be suitable for operation at relatively high speeds, for example as is necessary for driving of a drill bit, because relatively many and large parts with substantial moments of inertia must be reciprocatingly rotated and because several rotational parts with a mass distribution deviating from a rotationally symmetrical mass distribu-

tion are present.

The gear device of Utility Model No. 1 941 557 would further be unsuitable for the driving of a drill bit because its driven shaft executes reciprocating rotations having a rotational angle substantially smaller than 180° and because it occupies a relatively large space and is heavy. Moreover, the manufacture of this device is also relatively costly in view of the large number of required gears and toothed sectors, wherein costs are particularly increased by the fact that the tooth-free sector of the wheel on each of the intermediate shafts must have an edge projecting radially beyond the toothing and these wheels therefore cannot be constructed from commercially available gears.

United States patent specification No. 1 734 489 discloses a motor with two cylinders and pistons. Each piston is connected with a toothed rack engaging with a toothed sector of a wheel which also has two sectors with bevel toothings. A driven shaft carries two wheels with sectors having relatively offset bevel toothings, which shall alternately come into engagement with the bevel toothing of the first-mentioned wheel. This gear mechanism shall thus serve to convert reciprocating rotation - generated by the piston movements - of the first-mentioned wheel into unidirectional rotation of the driven shaft.

This gear mechanism thus does not serve for conversion of unidirectional rotation into reciprocating rotation. That apart, the toothed sectors of the two bevel gears on the driven shaft are certainly offset relative to each other. However, as far as can be ascertained, the toothings of the two bevel gears would exactly overlap if such

toothings were notionally continued around the whole gear circumference and projected one on the other. It seems feasible that the gear mechanism could not function otherwise. However, on each reversal of the rotational direction of the drive wheel, there then results, analogously to that previously described for the gear device of German Utility Model No. 1 941 557, a transient, entirely unengaged state of the bevel gear toothings and accordingly a high probability that the mechanism is blocked on reversal of the rotational direction of the drive wheel rotated by the toothed racks.

In German (Fed. Rep.) patent specification (Offenlegungsschrift) No. 29 36 004 there are disclosed rack and pinion switching mechanisms which each have a pair of gears for timed interengagement. The drive input gears have a toothed sector and a tooth-free sector, the radius of the tooth-free sector corresponding to the largest radius of the toothed sector. The drive output gears each have two circumferential positions at which teeth have been removed for the formation of recesses. These rack and pinion switching mechanisms apparently serve to convert unidirectional constant rotation of a drive shaft into intermittent, but still unidirectional, rotation of a driven shaft. These mechanisms thus do not make possible conversion of unidirectional rotation into reciprocating rotation.

There is therefore a need for a gear mechanism which serves for conversion of rotation in one direction into rotation alternately in mutually opposite directions and which avoids the disadvantages of the prior art mechanisms. Such a mechanism should especially avoid the risk of blocking that is inherent in the mechanism of German Utility Model No. 1 941 557, particularly when an element, such as a drill

bit, that exerts counter rotational moments on driven gears is to be driven. Moreover, the mechanism should preferably be capable of economic manufacture, and frictional losses, size and weight should be kept small, so the mechanism can be used with, for example, a surgical hand drill.

According to the present invention there is provided reversing rotary drive transmission means comprising two driven gears secure against relative rotation and a drive gear to drive the driven gears, either the drive gear or each of the driven gears being so provided with at least one tooth-free sector that on rotation of the drive gear in one direction the driven gears are alternately drivingly engaged by the drive gear for rotation alternately in mutually opposite directions, and the driven gears having the same tooth pitch spacing and being so displaced in phase relative to each other by an amount not exceeding one quarter of said spacing that the driven gear which at any one time is not drivingly engaged by the drive gear leads the other driven gear by that amount.

In such a transmission means, a fully unengaged state can be avoided through the phase displacement, within the specified limits, of the two driven gears and especially phase displacement in such a manner that the driven gear momentarily not in engagement with the drive gear leads the other driven gear. Through this leading of one driven gear by the other, on reversal of their direction of rotation the leading driven gear which will now come into engagement with the drive gear can prevent a free rotation of the drive gear. By this means it can be achieved that the driven gears can never be in a rotational setting in which the gear is blocked.

In a preferred embodiment of the invention, said amount of phase displacement is at least 5 percent, preferably 10 percent, more preferably 15 percent, of said spacing, and at most 23 percent, preferably 20 percent, of said spacing.

The transmission means according to the invention may be combined with translating means arranged to translate said rotation of the driven gears alternately in said mutually opposite directions into reciprocating rectilinear movement. This combination may be utilized in spooling apparatus, which further comprises a strand feed element driven by the translating means for reciprocating rectilinear movement, and rotatable spooling means so arranged relative to the feed element that a strand fed by the feed element to the spooling means during reciprocation of the former and rotation of the latter is spooled on the spooling means in alternately oppositely wound layers.

Embodiments of the present invention will now be more particular-ly described by way of example with reference to the accompanying drawings, in which:

Fig. 1 is a longitudinal sectional view of a gear transmission embodying the invention, which is attachable to a drilling machine or hand drill;

Fig. 2 is a partial plan view showing in greater detail one arrangement of gears in the gear mechanism;

Fig. 2A is a similar view showing another arrangement of gears in the gear mechanism;

Fig. 3 is a graph showing angular velocity of driven gears of the gear mechanism plotted against time, wherein the time periods for reversals in rotational direction

are exaggerated for the sake of clarity;

Fig. 4 is a diagram showing the pitch circles of the driven gears and the phase displacement of these gears;

Fig. 5 is an elevation of a hand drill having the gear mechanism of Fig. 1 operatively attached thereto; and

Fig. 6 is an elevation of spooling apparatus incorporating a gear mechanism embodying the invention.

Referring now to the drawings, and more particularly to Figs. 1 and 2, there is shown a gear mechanism 1 operative to convert unidirectional rotation into rotation in alternating directions. The mechanism 1 includes a housing 3 which is generally rotationally symmetrical about its longitudinal axis and which is basically composed of a cylindrical part shown to the right in Fig. 1 and a conical part shown toward the left, the conical part reducing towards one end of the housing.

The interior of the housing 3 is formed by a longitudinal opening 3a and a pair of mutually opposed radial openings 3b and 3c communicating with the longitudinal opening.

The longitudinal opening 3a is formed by means of a multiply stepped bore having a retaining ring 3 screw-fitted into the end section of the housing 3 remote from its conical portion. The retaining ring 5 has an external thread which detachably engages an internal thread formed in the housing 3. Additionally, the ring 5 is formed with fingers 5a which extend in the direction of the longitudinal axis of the housing 3 and which project away from the housing, the ring holding a ball bearing 7 inserted into the longitudinal opening 3a.

A rotatable element 9 is provided with a shaft journal 9a rotatably supported in the ball bearing 7 and with a bevel gear 9b. Furthermore, the element 9 is also provided at both ends with blind holes 9c and 9d, and a drive shaft 11 projects at one end thereof into the hole 9c, said end of the drive shaft having a radial recess 11a. The drive shaft 11 is connected with the element 9 at the shaft journal 9a by a pin 13 which extends through the recess 11a and which is inserted into appropriate openings formed in the shaft journal 9a.

A ring 15 is arranged on one end of the shaft journal 9a, which end projects out of the housing 3 and the retaining ring 5. The ring 15 is clamped to the drive shaft 11 by a clamping screw 17 which extends through a radial bore in the shaft journal 9a. The retaining ring 5 is sealed inwardly relative to the shaft journal 9b by a sealing ring 19.

A closure screw 21 and a sealing ring 23 are inserted into an opening 3b formed in the housing 3. A round disc 25 is inserted and rigidly fastened, for example by a pressed fit, in fluid-tight engagement in the outermost part of the opening 3c of the housing 3. A bearing bolt 27 extending outwardly from the housing interior is rigidly fastened, for example, by a pressed fit, in the disc 25 so as to be in fluid-tight engagement therewith. The end of the bearing bolt 27 projecting into the interior of the housing 3 is formed with a head 27a and a ring 29 is arranged on the interior side of the disc 25 with a sleeve 31 extending between the ring 29 and the head 27a of the bolt 27. The ring 29 and the sleeve 31 are composed of a self-lubricating material, for example with a metal basis, thereby enabling good sliding movement for rotation of the sleeve 31 around the

bolt 27.

Arranged on the sleeve 31 are bevel gears 33 and 35 which are rotatably fixed with one another by means of at least one pin 37 or the like, the bevel gears 33 and 35 being also rigidly fastened on the sleeve 31. The bevel gears 33 and 35 are rotatable about an axis 39 and together they form a rotatable element generally designated by reference numeral 41. The rotating element 41 will hereinafter be referred to as a drive element.

The bevel gear 33 meshes with the bevel gear 9b formed on the element 9. As will be seen from Fig. 2, the bevel gear 35 can have a toothed sector 35a which extends over only a part of the circumference of the bevel gear and a toothless sector 35b extending over the balance of the circumference, the toothed sector 35a extending over an angular range β. Of course, the toothless section 35b of the gear 35 extends over an angle of $360^{\circ} - \beta$.

A bush 43 is inserted in the blind hole 9d and a bush 45 is inserted in a part of the longitudinal opening 3a located in the area of the conical portion of the housing 3. The two bushes 43 and 45 are, for example, more or less fixedly pressed in place and they are composed of a self-lubricating material on a metal basis thereby enabling good sliding movement.

A shaft 49 having an axis 47 is rotatably supported in the two bushes 43 and 45, the axis 47 also forming the longitudinal axis of the housing 3 and of the bevel gear 9b and extending at right angles to the axis 39 of the pin 27.

Two bevel gears 51 and 53 are rotatably and axially fixed on the shaft 49 on axially opposite sides of the axis 39. The bevel gears 51

and 53 are detachably connected with the shaft 39 to be secure against rotation and axial displacement relative thereto. The gears 51 and 53 can be, for example, pressed on or possibly keyed to the shaft and the bevel gear 51 may be axially affixed relative to the shaft 49 by engagement between a shoulder formed on the shaft and the bush 43 and the bevel gear 53 by engagement between another shoulder on the shaft 49 and other auxiliary means.

Together with the bevel gears 51 and 53, the shaft 49 forms a rotatable element 55 which is the driven element of the mechanism and which will hereinafter be referred to as such. The two bevel gears 51 and 53 will henceforth be referred to as the driven bevel gears or driven gears and are formed with a toothed crown or gear ring extending over the entire periphery thereof, the gears 51 and 53 being arranged on opposite sides of the axis 39 in such a manner that during rotation the bevel gear 35 alternately achieves driving engagement with the bevel gears 51 and 53 in a manner to be described more fully hereinafter.

The longitudinal opening 3a is provided with an annular groove at the conical end of the housing 3 into which there is inserted a sealing ring 57 which seals the interior of the housing at the left-hand end of the shaft 49. A rubber collar 59 is arranged on the shaft 49 between the bush 45 and the bevel gear 53. The shaft 49 has a cylindrical blind hole 49a at the end thereof which projects out of the housing 3 and in the region of the collar 59 there is provided a radial hole with an entraining member 61 which projects into the blind hole 49a in order to enable a tool or other device to be held in operating position therein. In the area of the collar 59, the shaft 49

is provided with three other radial holes distributed uniformly over its circumference, with detent bodies 63, for example balls, being arranged in these holes, the detent bodies being resiliently pressed partially into the hole 49a by means of the collar 59. In addition, the collar 59 seals the hole containing the member 61 and the holes containing the bodies 63 and accordingly the hole 49a is also sealed against the interior of the housing.

In operation of the mechanism depicted in Figs. 1 and 2, the element 9 is driven in one rotational direction about the axis 47 and at, for example, a constant rotational speed.

The bevel gear 9b transmits the rotational movement of the element 9 to the bevel gear 33 and thereby also to the bevel gear 35 which, as previously described, is formed with a toothless sector 35b. The element 41 thus carries out rotational movement in only one direction, that is in the rotational direction designated by the arrow 65 in Fig. 2. Since the toothed sector 35a of the bevel gear 35 alternately engages the two bevel gears 51 and 53, the rotating element 55 formed by the shaft 49 and the bevel gears 51, 53 alternately rotates in different rotational directions.

In Fig. 3 the angular velocity ω of the element 55 is plotted against time t. The rectangular line 67 in the graph of Fig. 3 shows somewhat schematically the angular velocity ω of the driven element 55 in the case where the drive element 41 is driven at a constant angular velocity. A positive value of angular velocity of the element 55 occurs when the toothed sector 35a of the bevel gear 35 engages with the bevel gear 51. A negative value of angular velocity occurs when the toothed sector of the gear 35 engages the bevel gear 53. Of

course, it will be apparent that positive and negative values are arbitrarily selected.

It is assumed that the driven element 55 changes its rotational direction precisely at the point in time $t = 0$ and is stationary for a very short period of time, after which the toothed sector 35a of the bevel gear 35 engages the bevel gear 51. The angular velocity of the driven element 55 then increases abruptly to its positive maximum value and retains this velocity during a time period interval T_1. When the toothed sector 35a of the bevel gear 35 is disengaged from the bevel gear 51, the angular velocity of the driven element 55 abruptly decreases to 0 and retains this stationary condition for a time period T_2. When the teeth of the toothed sector 35a engage with the teeth of the bevel gear 53, the angular velocity abruptly achieves it maximum negative value and retains this value during a time period interval T_3. Subsequently, the driven element 55 is again stationary for a period of time T_4, after which a new cycle begins. The time period of a full revolution of the bevel gear 35 and a full period of the driven element 55 is designated by T.

The reversing gear mechanism 1 thus makes it possible to convert the unidirectional rotation of the drive element 41 into reversing rotation of the driven element 55. This action occurs in such a manner that the relationship between the angular velocities of the drive and driven elements 41 and 55 is practically constant during the entire backward and forward rotation of the element 55. The time period T_1 is equal to the time period T_3 and the time period T_2 is equal to time period T_4, the period T_2 and T_4 being represented with exaggerated magnitudes for the sake of clarity in the graph of Fig. 3.

At a constant angular velocity of the element 41, each time period T_1 and T_3 constitutes at least 48 percent, and preferably approximately 49 percent, of the total time T with the reversal time periods T_2 and T_4 each being approximately 1 percent of the total period T.

In order that the operational behaviour described above may be achieved and so that the reversing gear mechanism will operate properly, the bevel gears 35, 51 and 53 must fulfill certain specific conditions.

First, it is advisable that the teeth of the bevel gear 35 engage the teeth of the bevel gears 51 and 53 with only a small amount of play.

In addition, let it be assumed that z_{35} denotes the full, even number of gear teeth which the bevel gear 35 would have if the gear teeth were to extend over its entire circumference. Let the actual number of teeth provided in the sector 35a be designated by z_s and be a full, even or eventually uneven number. The pitch of the bevel gear 35 and of the bevel gears 51 and 53 is designated t_0. This represents the arc of the pitch circle which is between two right or left flanks of adjacent teeth. The term "engagement factor" or "overlapping degree" is designated ε and this factor is intended to mean the number of teeth of the gear 35 which in each instance engage with one of the gears 51 and 53. The value of the engagement factor ε rounded off to the next whole number is designated as ε. The engagement factor ε is usually equal at least to 1.3 and for the reversing gear mechanism it is preferred if this factor is at least 1.5 but less than 2.

Thus, the number of teeth in the sector 35a is given by the following equation, in which ε preferably has the value 2:

$$z_8 = (z_{35}/2) - E \qquad (1).$$

The two bevel gears 51 and 53, or more precisely their toothings, are substantially identical and have the same pitch circle, the same inner and outer diameters and the same number of teeth, this number of teeth being even or odd. The bevel gears 51 and 53 are arranged on the shaft 49 in such a way that their teeth are somewhat rotationally offset relative to one another when one gear is projected on the other in a direction parallel to the axis 47 or when they are notionally overlapped by means of non-rotating axial displacement. This rotational offset which is also indicated in Fig. 2 is explained in greater detail with reference to Fig. 4. In Fig. 4 there are shown the axes 39 and 47, a pitch circle 75 of the bevel gear 35, a pitch circle 81 of the bevel gear 51 and a pitch circle 83 of the bevel gear 53 with the centres of the pitch circles also being shown. Moreover, a position or point 85 and 87 is marked respectively on each of the two pitch circles 81 and 83. These points or positions designate mutually corresponding locations of a pair of teeth on the opposed gears, which locations are on the pitch circles, for example points of intersection of the tooth centre lines with the pitch circles. Moreover, if straight lines 89 and 91 are drawn parallel to the axis 47 it will be noted that these lines will lie in different planes extending through the axis 47 and that they intersect the two pitch circles 81 and 83. The line 89 extends through the position 85 on the pitch circle 81 and the line 91 extends through the position 87 on the pitch circle 83. The two straight lines 89 and 91 thus represent projections of the positions 85 and 87, the position 85 being projected onto the pitch circle 83 and the position 87 being projected

onto the pitch circle 81. It will be noted from Fig.4 that the two positions 85 and 87 are angularly offset relative to each other by a distance represented by an arc a measured on the pitch circle 81 or 83 and by a central angle α measured in the centres of the two pitch circles 81 and 83 respectively, which angle will also be designated as a phase angle in the following description. The arc a thus represents an amount of phase displacement of the gears 51 and 53 relative to each other.

In Fig. 4 there is shown a further line 93 which connects the two positions 85 and 87 and which extends along a notional cylindrical surface containing the two pitch circles and thereby has the form of a part of a helix. The line 93 extends in the manner of a right-handed thread. The bevel gear 35 rotates during operation in the direction of the arrow 65, which taken in a viewing direction seen from the intersecting point of the axes 39 and 47, corresponds to a clockwise rotational movement. Moreover, when viewing the bevel gear 53 from the aforementioned intersection point, the position 87 is offset, i.e. rotated about the axis 47, counterclockwise by the arc a relative to the position 85. When viewing the bevel gear 51 from the intersecting point, the position 85 of the bevel gear 51 is accordingly offset in a counterclockwise direction through the arc a relative to the projection of the position 87.

It should be noted here that the gear 35 can also be rotated in the opposite direction whereby the bevel gears 51 and 53 would then be offset relative to one another in such a manner that the line corresponding to the line 93 would extend in the manner of a left-handed thread.

In operation, the toothing of that bevel gear 51 or 53 which momentarily is not engaged with the driving gear wheel 35 is, in each instance, displaced forwardly in accordance with the phase angle, i.e. leading in the instantaneous running direction by the arc a relative to the other bevel gear 51 or 53, respectively. The length of the arc a is at most equal to $t_0/4$ or preferably smaller than $t_0/4$. The length of the arc a is therefore between the values 0 and $t_0/4$ whereby the latter value itself can eventually also belong to the interval so that the following equation applies:

$$0 < |\underline{a}| \leq t_0/4 \qquad (2).$$

The arc a may be at least 5 percent, preferably at least 10 percent and more preferably at least 15 percent, of the tooth pitch t_0 and it may be, for example, at most 23 percent, preferably at most 20 percent, of the pitch t_0.

When the reversing gear mechanism is constructed in the manner described above, particularly if the requirements expressed in the formulae (1) and (2) are fulfilled, and if the bevel gears 35, 51 and 53 have a normal tooth flank play, the driven element 55 cannot freely rotate by itself in any rotational position of the drive element 41. The bevel gears 35, 51 and 53 can then also never achieve a rotational position relative to one another where the mechanism is blocked, i. e. is jammed. Naturally, the result of the tooth displacement or offset is that the drive element 41 can be driven only in one rotational direction. Should a drive in the reverse direction be desired, the relative positions of the driven gears must be adjused.

As already mentioned, the two bevel gears 51 and 53 have the same number of teeth $z_{51,53}$. The amount or absolute value of the positive or negative rotational angle ϕ through which the driven element 55 is alternately moved is given by means of the following

equation, where o^{0} is the rotational angle measured in degrees:

$$|\phi^{o}| = 180\ z_{35}/z_{51,53} \qquad (3).$$

The gear mechanism 1 may be utilized for driving a twist or spiral drill bit, for example, and it may be detachably connected to a drilling machine or hand drill. In Fig. 5 there is shown a drilling device which comprises a hand drill 101 provided for use in bone surgery with a housing 103, a compressed air connection 105, a compressed air motor and at least one manually movable actuation mechanism 107. With the latter, the motor may be energized and de-energized and the number of motor revolutions may preferably be changed in a continuous manner. The drive shaft 11 shown in Fig. 1 may be formed by means of the shaft of the hand drill 101, which shaft may also be used for holding a detachable drill chuck. Moreover, the fingers 5a are constructed so that they engage in recesses of the drill housing 103 and can thereby connect, in relatively fixed rotative engagement, the housing 3 with the housing 103 of the hand drill 101.

A twist drill bit 111 having a cutting part with two cutting edges at the forward end thereof and having a shaft which can be inserted into the hole 49a so as to be detachably connected with the shaft 49 is shown in Fig. 5. In order to achieve detachable connection, the generally cylindrical drill shank is provided with an annular groove in the region of its end in which the three detent bodies 63 can engage. Additionally, the drill shank may have a flattened portion at which the entraining member 61 may engage thereby connecting the shaft 49 with the drill bit 111 in mutually rotating engagement.

Each of the cutting edges is rotated alternately in opposed directions through a sector having central angle equal to the rotational angle ϕ. The rotational angle should therefore be at least equal to or preferably larger than 180° so that the two sectors just mentioned will overlap. On the other hand it is advisable for drilling that the rotational angle ϕ be at most 360° and preferably less than 360°. The rotational angle ϕ can, for example, be within the range between 200° and 300°.

The desired operation parameters of the invention may be accomplished in accordance with formula (3) by determining the corresponding number of teeth for the gears 35, 51 and 53. For example, if $z_{35} = 24$ teeth, $z_s = 10$ teeth and $z_{51,53} = 16$ teeth, then the angle β would be approximately 150° and the rotational angle ϕ would be approximately 270°. The engagement factor ε may have a magnitude of, for example, 1.63 so that $E = 2$. The arc a may then be about 0.18 t_0.

The parts rotating backwards and forward in operation, namely at least the driven element 55 and the drill bit 111 held by this, have only a relatively small mass and also only a small moment of inertia. In the case of the bevel gear 35 in essence only the teeth are directly removed in the tooth-free sector 35b, while the inner part of the gear is still present in this sector. However, the mass distribution of the gear 35 thereby departs only relatively slightly from a rotationally symmetrical distribution and the remaining parts rotating on operation are virtually completely rotationally symmetrical. It is therefore readily possible to use the gear mechanism for conversion of relatively high-speed rotational movement, which results in a favourable cutting speed of the drill bit 111. The drive element

41 with the gear 35 can be, for example, rotated at a speed, according to the drill bit diameter, which is at least about 250, preferably about 400, r.p.m. and at most 750 r.p.m., whereby the gears 51 and 53 and drill bit connected thereto perform the same number of revolutions in both directions.

During drilling of a hole, the angular velocity of the drill bit 111 will be the same as the angular velocity ω of driven element 55 and it will thus be maintained constant during the periods t_1 and t_3. This makes it possible to adjust the r.p.m. of the drill in such a way that in each instance it will have a favourable cutting speed during the entire rotational movement during which cutting occurs.

When a hole is drilled in a bone during a surgical procedure, it may occur that after drilling through the bone, the drill may enter soft tissue and tear such tissue loose when the drill tip exits from the bone. The backward and forward rotations of the drill bit 111 prevent winding about the drill and consequent tearing of soft tissue parts.

Since the movable parts of the mechanism 1 necessary for converting rotational movement are all rotatably supported and formed by means of shafts and toothed gears, the efficiency of power transmission in the conversion is quite favourable. Moreover, the mechanism 1 may be constructed so as to be small and light relative to its output performance so that the work of the surgeon during the use of the drilling device shown in Fig. 5 will not be impeded. The bevel gear 35, 51 and 53 can be so dimensioned that their largest external diameter is at most 25 millimetres. If, for example, the modulus is

0.75 millimetres and the tooth numbers have the aforesaid values, this results in a pitch circle diameter of 18 millimetres for the gear 35 and 12 millimetres for each of the gears 51 and 53. The external diameter of the gear 35 can then be 19.5 millimetres and that of each of the gears 51 and 53 13.5 millimetres. Moreover, the modulus and tooth number can naturally be varied, although the tooth number z_{35} in the case of a gear mechanism to be mounted on a hand drill is preferably at most about 36. Since all the bearings and gears are housed in the interior of the housing 3 which is sealed and closed off from its surroundings, the mechanism can also be sterilized and be kept sterile with little expense. If necessary, the interior of the housing 1 may be made accessible by removing the sealing or closure screw 21.

Another application of the mechanism of the invention is depicted in Fig. 6 which shows a spool winding device provided with a frame and with means for detachably supporting a rotatable spool or bobbin 151, which is mounted on a shaft 155 rotatably supported at one end by a bearing 153 and driven by a drive mechanism 157 comprising, for example, an electric motor. A mechanism 161 having a housing fastened to the frame of the winding device includes a reversing gear mechanism for converting unidirectional rotation into bidirectional alternating rotation. The reversing gear mechanism has three bevel gears corresponding to the bevel gears 35, 51 and 53, with the bevel gear corresponding to the gear 35 being arranged on a drive shaft 163. The drive shaft 163 is connected with the drive mechanism 157 by means of a gear unit 165 through replaceable toothed gear wheels. The driven shaft 167 of the mechanism 161 is operatively connected with a wheel

171 through a gear unit 169 which may be, for example, a spur gear or a worm gear. The wheel 171 is rotatably supported about an axis which extends perpendicularly to the rotational axis of the spool 151. The wheel 171 is drivingly coupled to another rotatable wheel 173 by way of a belt 175 extending parallel to the axis of rotation of the spool 151. A guide mechanism 177 is fastened to the belt 175 and serves to guide a flexible element 179 which is to be wound about the spool 151. The flexible element may be, for example, an insulated electrical wire.

When a wire or similar strand of material is wound on the spool 151 during operation of the winding device, the drive mechanism 157 rotates the shaft 155 carrying the spool 151 as well as the driving shaft 163 of the mechanism 161. The drive shaft 163 is rotated with a unidirectional rotational motion and it may be rotated at a constant or variable speed. The driven shaft 167 of the mechanism 161 then will be driven with an alternating bidirectional rotational movement so that the belt 175 of the guiding mechanism 177 will be moved in a reciprocating fashion indicated by the arrow 181 in directions parallel to the rotational axis of the spool. The displacement speed of the guiding mechanism is determined in such a manner that the windings of the element 179 will lie adjacent one another without intervening spaces in each layer of the coil that is formed. Furthermore, the displacing distance of the guiding mechanism 177 is equivalent to the length of the spool so that in each instance the guide means changes direction after winding of a single layer on the spool. It is of significant advantage that the guide mechanism 177 changes direction very quickly as indicated from Fig. 3 and will accordingly

practically always produce the same winding characteristics during each backward and forward movement.

The mechanisms 1 and 161 can be manufactured at relatively low cost. In this connection it is of particular significance that bevel gears of usual construction can be used for the mechanism and for the formation of the tooth-free sector in the case of the drive bevel gear it is only necessary to effect a corresponding tooth removal by milling or by other means before or after manufacture of the toothing.

The mechanisms 1 and 161 for converting the rotational movement may be modified in various respects. For example, the two bevel gears 51 and 53 can be formed together with the shaft 49 as a unitary body. Moreover, the bevel gear 35 may have another uneven number of toothed and tooth-free sectors, for example three uniformly distributed sectors with teeth thereon and three toothless sectors therebetween, in place of the two sectors 35a and 35b. Then, instead of formula (1) applying, the following more general formula would apply:

$$z_s = (z_{35}/2n) - E \qquad (4)$$

wherein n designates the number of toothed sectors of the gear wheel 35. The remaining symbols have the same meaning as in formula (1), with E again preferably being 2. Furtheron, z_{35} must of course be divisible by 2n so that z_s is a full, even or uneven number.

As previously indicated in accordance with the preferred embodiments, n is an odd integer, preferably equal to 1 or 3. In an embodiment shown in Fig. 2A wherein n = 3, there is provided a gear wheel 35' having three toothed sectors 35a' and three toothless sectors 35b'. The sectors 35a' each extend over an angle β' which is, for instance, 30° and the toothless sectors 35b' each extend over an angular

distance of, for instance, 90^{0}.

In this embodiment, $z_{s} = 2$ so that each toothed sector 35a' has two teeth. The driven gears 51, 53 are each provided with sixteen teeth. However, gears with more teeth could also be provided.

Moreover, the drive gear can have a full toothing and each driven gear at least one tooth-free sector. The toothed sectors of the driven gears would then be offset opposite to each other, so as to be alternately engaged by the drive gear . If the sectorial toothings of the driven gears are notionally supplemented to provide full toothings, these notional additional teeth would be similarly oppositely offset or rotated, as can be explained for the gears 51 and 53 on the basis of Fig. 4. Instead of the variant with sectorially toothed driven wheels of which the sectorial toothings are to be notionally supplemented to complete toothings, there can also be considered the offsetting of two desired mutually corresponding toothing positions, for example tooth centre lines of two desired teeth, of the two driven gears. If these toothing positions are offset by an arc b measured on the notionally superimposed pitch circles of the driven gears, this arc should differ from the pitch t_0 by an integral multiple and should be greater or smaller by an arc a than the integral multiple of t_0 that is most similar in magnitude, wherein a lies in the magnitude range defined by the formula (2). In other words, a is the same as the difference between b and that integral multiple of t_0 from which b differs the least.

Further, in the case of the gear mechanism 1 there can be provided an externally manually adjustable adjusting element and a sleeve slidable with the adjusting element along the collar 59, the

sleeve serving to selectably block or release the detent bodies 63 in the state engaging in the drill bit shank. With such a variant the drill bit engagement can be selectably blocked or released by rotation of the adjusting and locking element.

In the case of the gear mechanism 1, the fastening means serving forfastening to a drilling machine can be modified according to the construction of the particular machine. For example, the fingers 5a of the holding ring 5 can be replaced by a clamping sleeve able to be firmly clamped to the machine housing, in which case the clamping screw 17 can be omitted and the clamping connection between the rotatable element 9 and the drilling machine shaft can be replaced by a pure plug connection.

Moreover, the pressed-in disc 25 can be replaced by a disc which is releasably fastened in place and sealed off by a sealing ring, in which case the opening 3b of the housing, the screw 21 serving for the closure thereof, and the sealing ring can be omitted.

The embodiments of the invention in which an exclusive property or privilege is claimed are defined as follows:

1. Reversing rotary drive transmission means comprising two driven gears secure against relative rotation and a drive gear to drive the driven gears, either the drive gear or each of the driven gears being so provided with at least one tooth-free sector that on rotation of the drive gear in one direction the driven gears are alternately drivingly engaged by the drive gear for rotation alternately in mutually opposite directions, and the driven gears having the same tooth pitch spacing and being so displaced in phase relative to each other by an amount not exceeding one quarter of said spacing that the driven gear which at any one time is not drivingly engaged by the drive gear leads the other driven gear by that amount.

2. Transmission means as claimed in claim 1, wherein the driven gears have the same number of teeth and same pitch circle diameter, and the number of tooth-free sectors provided on the drive gear or each driven gear is an odd number.

3. Transmission means as claimed in claim 2, wherein the odd number is at most three.

4. Transmission means as claimed in any one of claims 1 to 3, wherein either the drive gear or each driven gear is provided with only one such tooth-free sector and with only one toothed sector.

5. Transmission means as claimed in any one of claims 1 to 3, wherein the drive gear is provided with at least one tooth-free sector and the toothing of each of the driven gears is continuous.

6. Transmission means as claimed in any one of claims 1 to 3, wherein the actual number of teeth in the or each toothed sector of the or each gear provided with said at least one tooth-free sector

is equal to $\frac{Z_{35}}{2n} - E$, wherein Z_{35} is the number of teeth in a given notional full complement of teeth for that gear, n is the number of toothed sectors of that gear and E is the number, rounded off to the next higher whole number, of teeth of that gear which are simultaneously engaged for transmission of drive thereto or therefrom.

7. Transmission means as claimed in any one of claims 1 to 3, wherein said amount of phase displacement is at least five percent of said spacing.

8. Transmission means as claimed in any one of claims 1 to 3, wherein said amount of phase displacement is at least 10 percent of said spacing.

9. Transmission means as claimed in any one of claims 1 to 3, wherein said amount of phase displacement is at least 15 percent of said spacing.

10. Transmission means as claimed in any one of claims 1 to 3, wherein said amount of phase displacement is at most 23 percent of said spacing.

11. Transmission means as claimed in any one of claims 1 to 3, wherein said amount of phase displacement is at most 20 percent of said spacing.

12. Transmission means as claimed in any one of claims 1 to 3, wherein the drive gear is provided with at least one tooth-free sector, and the number of teeth in a notional full complement of teeth for the drive gear is an even number.

13. Transmission means as claimed in any one of claims 1 to 3, wherein each of the driven gears is provided with at least one tooth-free sector and the drive gear has an even number of teeth.

14. Transmission means as claimed in any one of claims 1 to 3,

A

wherein the drive gear is provided with at least one tooth-free sector and the or each toothed sector of the drive gear has an even number of teeth.

15. Transmission means as claimed in claim 1, wherein the gears are bevel gears and the driven gears are mounted on a rotational drive shaft to be secure against relative rotation and are displaced in phase relative to each other by said amount about the rotational axis of the shaft.

16. Transmission means as claimed in claim 15, wherein the driven shaft is provided at one side of the two driven gears with fastening means for detachable fastening to the shaft of a drill bit and the transmission means further comprises a fourth bevel gear arranged co-axially with the drive gear for rotation therewith, and a fifth bevel gear disposed at the other side of the two driven gears and drivably engaging the fourth bevel gear, the fifth bevel gear being rotatable about the same axis as the driven shaft and being releasably and drivingly couplable with a drive shaft of a prime mover.

17. Transmission means as claimed in claim 1, wherein the teeth of the drive and drive gears are so arranged that each of the driven gears is driven by the drive gear when engaged therewith through a rotational angle of at least 180°.

18. Transmission means as claimed in claim 17, wherein the rotational angle is in the range of 200° to 300°.

19. Transmission means as claimed in any one of claims 1 to 3, wherein each of the drive and driven gears has an outer diameter of at most 25 millimetres and has at most 36 teeth.

20. Transmission means as claimed in claim 1, wherein said amount of phase displacement is greater than zero and smaller than one quarter of said spacing.

21. Transmission means as claimed in claim 1 in combination with translating means arranged to translate said rotation of the driven gears alternately in said mutually opposite directions into reciprocating rectilinear movement.

22. Spooling apparatus comprising the combination of transmission means and translating means claimed in claim 21, a strand feed element driven by the translating means for reciprocating rectilinear movement, and rotatable spooling means so arranged relative to the feed element that a strand fed by the feed element to the spooling means during reciprocation of the former and rotation of the latter is spooled on the spooling means in alternately oppositely wound layers.

1205654

3-1

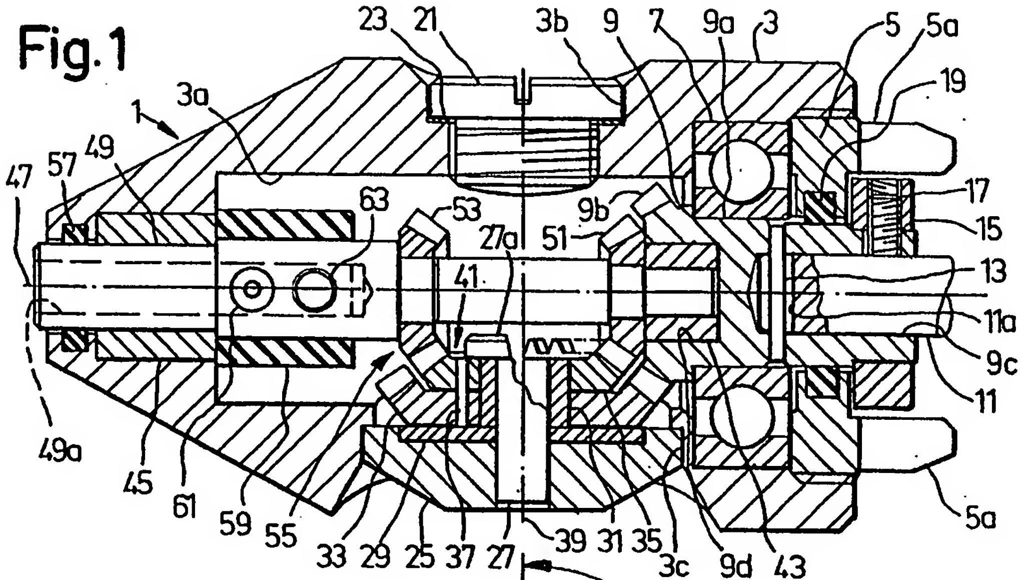

Fig. 1

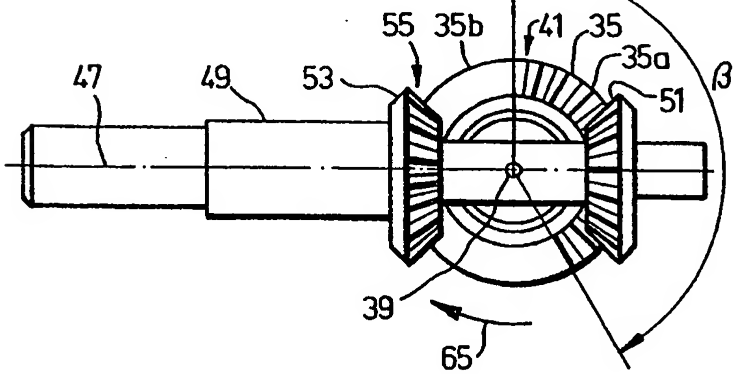

Fig. 2

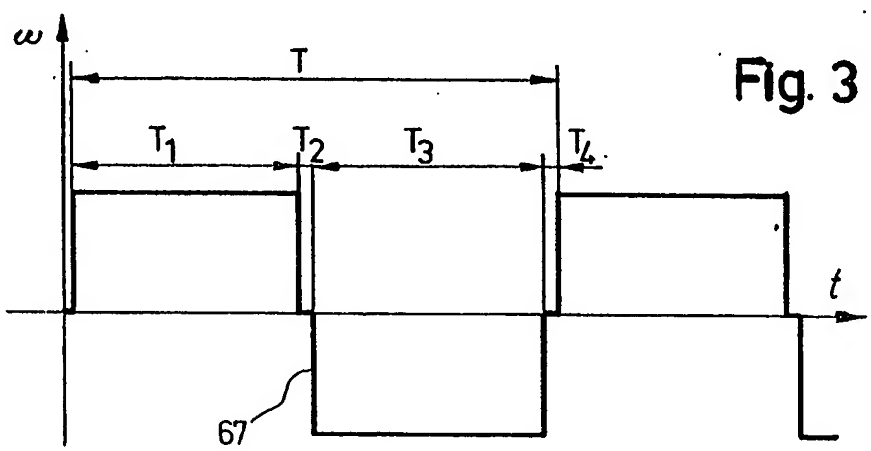

Fig. 3

Patent Agents

Swabey Mitchell Houle Marcoux & Sher

1205654
3-2

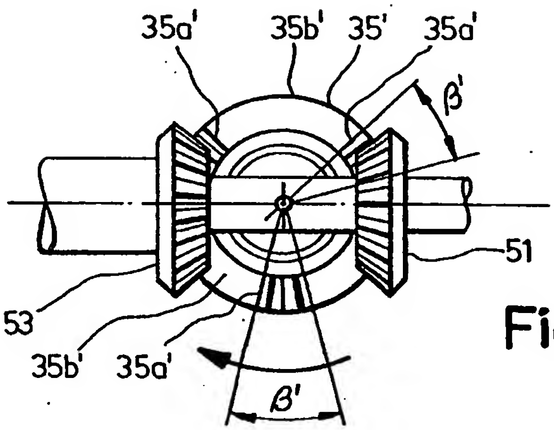

Fig. 2A

Patent Agents
Swabey Mitchell Houle Marcoux & Sher

1205654
3-3

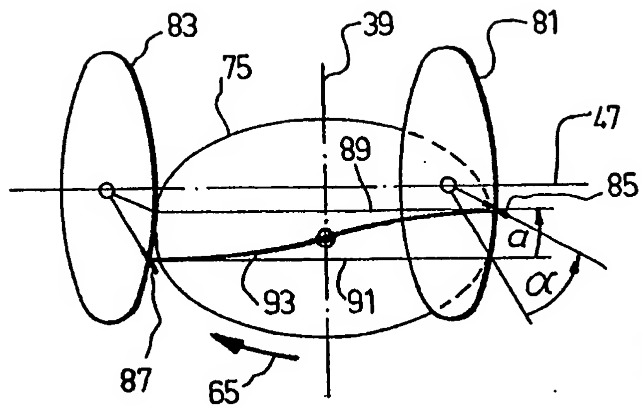

Fig. 4

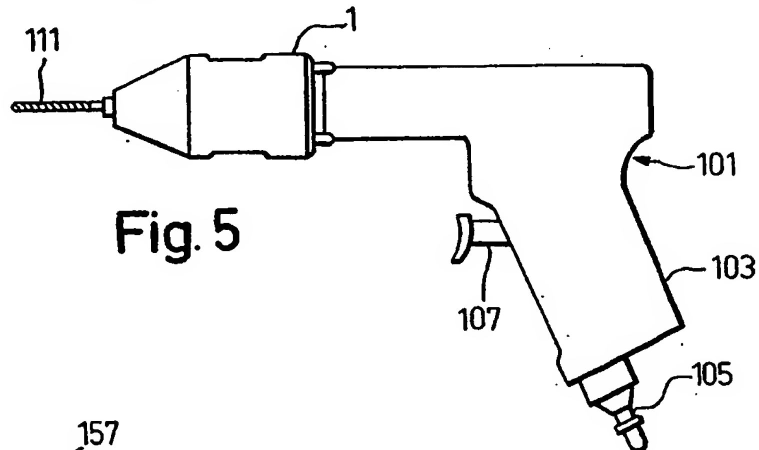

Fig. 5

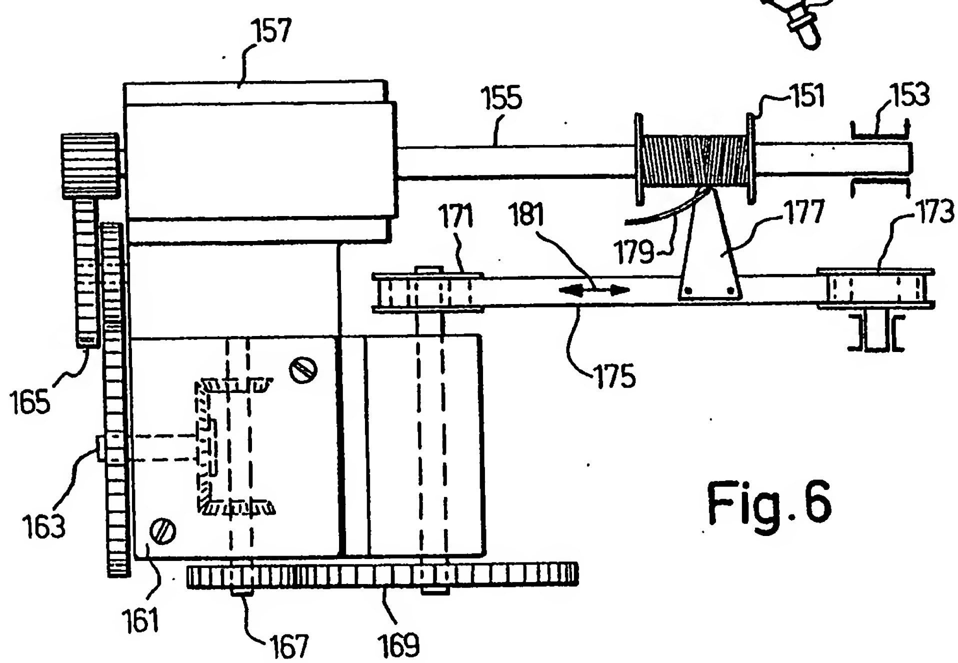

Fig. 6

Patent Agents
Sabey Mitchell Houle Marcus & Ahern